Axial piston compressor, especially a compressor for the air-conditioning system of a motor vehicle

Description

The invention relates to an axial piston compressor, especially a compressor for the air-conditioning system of a motor vehicle, having a housing and, for drawing in and compressing a coolant, a compressor unit arranged in the housing and driven by means of a drive shaft, the compressor unit comprising pistons moving axially back and forth in a cylinder block and comprising a swash plate which drives the pistons and rotates together with the drive shaft.

An axial piston compressor of such a kind is known, for example, from DE 197 49 727 A1. That compressor comprises a housing in which, in a circular arrangement, a plurality of axial pistons are arranged around a rotating drive shaft. The drive force is transmitted from the drive shaft, by way of an engaging member, to an annular swash plate and in turn, from there, to the pistons displaceable in translation parallel to the drive shaft. The annular swash plate is pivotally mounted on a sleeve which is mounted on the drive shaft so as to be axially displaceable. In the sleeve there is provided an elongate hole, through which the mentioned engaging member engages. Consequently, the capability of the sleeve for axial movement on the drive shaft is limited by the dimensions of the elongate hole. Assembly is carried out by passing the engaging member through the elongate hole. The drive shaft, engaging member, sliding sleeve and swash plate are arranged in a so-called drive mechanism space, in which gaseous working medium of the compressor is present at a particular pressure. The delivery volume and consequently the delivery performance of the compressor are dependent on the pressure ratio between the suction side and delivery side of the pistons or correspondingly dependent on the pressures in the cylinders on the one hand and in the drive mechanism space on the other hand.

A somewhat different kind of construction of an axial piston compressor is described, for example, in DE 198 39 914 A1. The swash plate is in the form of a wobble plate, there being arranged between the wobble plate and the pistons a non-rotating take-up plate mounted opposite the wobble plate.

Reference is made, furthermore, to the following prior art:

DE 2 524 148
US 4 815 358
US 4 836 090
US 4 077 269
US 5 105 728

In the case of the compressors described in those publications, the purpose is, *inter alia*, to take measures to prevent or reduce drive mechanism imbalance in use. Otherwise the known arrangements have in common the fact that the rotating components are of relatively large and, consequently, heavy construction compared to the parts moved in translation, namely the pistons, piston rod etc.. Furthermore, the known arrangements have in common the fact that the actual swash plate apparatus is acted upon by an additional plate by means of a suitable coupling mechanism. The several rotating components are intended to bring about a righting moment of the swash plate apparatus in the direction of minimum stroke, which has an influence on the regulatory behaviour.

The mentioned arrangements are all relatively complicated, expensive and of low compactness and for that reason they are unsuitable for the compressors required nowadays by the automobile industry for air-conditioning systems.

Also in the case of serially produced compressors as are used nowadays in motor vehicles, it is an objective that the components moved or masses moved should be suitably dimensioned in order to achieve a desired regulatory behaviour, more particularly so that the centrifugal forces occurring on rotation of the swash plate are sufficient to counteract the pivoting movement so as to provide deliberate regulation and thereby to influence, especially to reduce or to limit, the piston stroke and, accordingly, the quantity delivered. The compressor 6SEU12C serially produced by DENSO has, for example, a drive mechanism having the following masses relevant to the regulatory behaviour:

Component	Number	Mass of component [g]	Total mass [g]
Pistons	6	41	246
Sliding block	12	5	60
Masses moved in translation			**306 g**
Swash plate	1	391	391
Guide pins	2	20	40
Masses moved in rotation			**431 g**

From the above-mentioned figures it can be seen that a considerable component mass is provided for the parts moved in rotation. By that means an attempt is made to produce a sufficient counter-force or counter-moment relative to the masses moved in translation. The same basic idea also underlies DE 198 39 914 A1, in which indeed the rotating mass of the swash plate or of the pivotal part thereof is so dimensioned that the centrifugal forces occurring on rotation of the drive plate are sufficient to counteract the pivotal movement of the swash plate to provide deliberate regulation and consequently to influence, namely to reduce or to limit, or especially to keep constant, the piston stroke and accordingly the quantity delivered.

The publication by Björn Fagerli "A theoretical comparison of the mechanical control behaviour of a R744 and a R134a automotive AC compressor", published in the context of the Purdue Compressor Conference 2002, describes the influencing variables which act as moments about the centre of tilt of a swash plate apparatus. In detail, these are the following moments, the particular direction of the moments being given in brackets and with (-) denoting down-regulation (in the direction of minimum stroke) and (+) denoting up-regulation (in the direction of maximum stroke):

- moment due to gas forces in the cylinder spaces (+)
- moment due to gas forces from the drive mechanism space (-)
- moment due to a restoring spring (-)
- moment due to an advancing spring (+)
- moment due to rotating masses (-); including moment due to location of centre of gravity (for example, swash plate: tilt position ≠ mass centre of gravity): can be (+)
- moment due to masses moved in translation (+)

In relation to the mentioned 6SEU12C compressor of DENSO, which represents the typical constructional form of a swash plate compressor, it is to be noted that the mass of such a swash plate cannot be increased at will in order to modify the regulatory behaviour accordingly. This is due to the fact that, in the case of the compressors of the described kind, the mass centre of gravity of the swash plate is generally a substantial distance away from the tilt-providing articulation of the swash plate. The basic justification for such an arrangement is that the swash plate, in addition to its own guideway on the drive shaft, has to be coupled to the drive shaft or a component connected to the drive shaft by way of a positioning mechanism.

The mentioned distance between the centre of gravity of the swash plate and the tilt-providing articulation thereof results in imbalance of the drive mechanism, especially in dependence upon the swash plate tilting angle, and in the worst case results in an up-regulating characteristic (see above "location of centre of gravity").

Accordingly, in the case of the compressors according to the prior art, and indeed according to both the published and the actually practised prior art, a compromise has to be reached so that a predetermined mass of the swash plate is made available in order to produce a counter-moment to the masses moved in translation; on the other hand, however, the mass of the swash plate must not be over-dimensioned because then the imbalance of the drive mechanism would be excessive.

In order to address that problem it has also already been proposed that the pistons, that is to say the masses moved in translation, should be constructed as sparingly, that is to say as lightly, as possible, for example using aluminium or other materials of relatively low specific density. In that respect it has also been proposed to use hollow pistons.

However, even using those measures, it is not possible to achieve, in particular, constant regulation of the quantity delivered at different speeds of rotation. In this context it should be pointed out that the phrase "constant regulation" is not to be understood as an exact statement. The quantity delivered would be exactly constant only if, for example, on doubling the speed of rotation, the tilting angle of the swash plate apparatus halves. It has to be borne in mind, however, that other parameters besides affect the quantity delivered, for example volumetric efficiency or oil throw-off or the like, when, for example, the tilting angle of the swash plate changes.

For constant regulation of the quantity delivered in the event of changing speeds of rotation, the restoring torque of the swash plate apparatus is utilised because - as already explained - the swash plate counteracts its angled position because of the dynamic forces at the co-rotating plate part.

This process can be aided by spring forces or by hydraulic, pneumatic or like means so that, in the case of an increase in the speed of rotation, an increasing quantity delivered is compensated by restoration of the angled position, at least in part.

The case of compensation of the up- and down-regulating moment can likewise be very interesting. Changes in the speed of rotation then do not influence the regulation. Accordingly it is possible to proceed with a relatively simple regulation algorithm.

As already mentioned hereinbefore, such a behaviour can in principle be obtained by, for example, integrating an additional mass into the drive mechanism, the inertia of which mass acts, as described, on the swash plate apparatus by way of a coupling mechanism.

However, it has also already been explained that the mass of the swash plate cannot be increased at will without having to accept other disadvantages. This also holds true, especially, for the teaching according to DE 198 39 914 A1 and EP 99 953 619 (Application No.). The regulation proposed therein using the mass of the rotating components may result in regulatory behaviour as a result of which the delivery output is substantially independent of the speed of rotation but this is not necessarily the case. Over-compensation may also be an outcome. The dimensioning criteria are blurred. The reason for that lies in the fact that the mass of the rotating components influences the righting moment of the swash plate only proportionally.

The problem underlying the present invention is accordingly to provide a compressor of the kind mentioned at the beginning which has, as desired, a regulatory behaviour of compensation, over-compensation or constant regulation for the "quantity delivered", that being the case, more particularly, using a minimum mass of pivotal rotating components so that it is possible for the compressor to have a compact form of construction.

The problem is solved in accordance with the invention by the characterising features of claim 1, preferred constructional details and further developments being described in the subordinate claims.

Accordingly, the desired regulatory behaviour of the compressor is primarily achieved, in accordance with the invention, not using the component mass but rather by taking into account the mass moment of inertia of the swash plate arrangement, the former being dependent on the geometry of the latter.

A central idea of the invention is accordingly to compensate, or even to over-

compensate, the moment due to translational masses directly by the moment due to rotating masses.

The righting moment to be brought about at a swash plate apparatus is a function of the speed of rotation or angular velocity ω and the mass moment of inertia J of the swash plate apparatus:

$M = f(\omega^2; J)$

The mass moment of inertia itself is substantially a function of the component mass and the component geometry, defined, for example in the case of a disc, by the diameter "2r" and the disc thickness or height "h":

$J = f(m, r^2, h^2)$

In even more precise terms, the mass moment of inertia is substantially a function of the component density distribution and of the component geometry, component density distribution taking into account, for example, swash plates made of disparate materials, namely 2, 3 or more materials, or made of one material having a varying density distribution (metal foam, heterogeneous material):

$J = f(\varrho, r^2, h^2)$
wherein ϱ = density,
r = swash plate radius, and
h = swash plate height.

In addition, the location of the component centre of gravity has to be taken into account. Preference is given to a component centre of gravity on the axis of the drive shaft, especially at the tilting point of the swash plate apparatus (that is to say, then, for each particular tilting angle).

From the contexts it will be seen that it is effective (exponent) to select the geometry of the swash plate apparatus so that the desired regulatory behaviour is achieved.

It is especially advantageous if there is provided a swash plate geometry which represents a compromise between "low component mass" and (sufficiently) "high mass moment of inertia".

When the swash plate is in the form of a swash ring, this can be achieved by maximally dimensioning both the inner diameter and also the outer diameter whilst taking into account the external environmental conditions, the external environmental conditions being governed by the size of the drive mechanism space and also, for example, by the necessary sliding and bearing surface for the sliding blocks of an articulation arrangement between the swash plate or swash plate ring and the pistons. It is also possible to influence the desired mass moment of inertia by suitably selecting the thickness of the swash plate.

An exemplifying embodiment of a swash plate drive mechanism according to the invention is explained in greater detail hereinbelow with reference to the accompanying drawings, in which:

Fig. 1 shows, in a diagrammatic perspective view, an arrangement of a swash plate mechanism, in accordance with the invention, for an axial piston compressor for vehicle air-conditioning systems, the swash plate being in a maximal piston position;

Fig. 2 shows the mechanism according to Fig. 1 in a diagrammatic side view;

Fig. 3 shows the swash plate mechanism according to Figs. 1 and 2, partly in a side view and partly in cross-section;

Fig. 4 shows the swash plate mechanism according to Fig. 3 in a side view;

Fig. 5 shows the mechanism according to Figs. 1 - 4 in a diagrammatic perspective view, the swash plate being located in a minimal piston stroke position;

Fig. 6 shows the mechanism according to Fig. 5 in a side view;

Fig. 7 shows the swash plate mechanism according to Figs. 5 and 6, partly in a side view and partly in cross-section;

Fig. 8 shows the swash plate mechanism according to Fig. 7 in a side view;

Fig. 9 is a diagrammatic representation of the co-ordinates of a swash plate mechanism for calculating the mass moment of inertia; and

Fig. 10 shows, in cross-section and to an enlarged scale, part of a compound swash ring.

Figs. 1 - 8 show, in diagrammatic form, a preferred arrangement of a swash plate mechanism 100 for an axial piston compressor for motor vehicle air-conditioning systems. This swash plate mechanism 100 comprises a swash plate 107, which is modifiable in terms of its inclination relative to a drive shaft 104 and which is driven in rotation by the drive shaft and which in the present case is annular, the swash plate 107 being in articulated connection both with a sliding sleeve 108 mounted on the drive shaft 104 so as to be axially displaceable and also with a supporting element 109, which is spaced away from the drive shaft 104 and rotates together with the latter. This articulated connection is in the form of axial support, as can be seen especially well from Figures 2 - 4 and 5 - 8. The co-operation of the swash ring 107 with axial pistons which are arranged evenly distributed over a circumference extending around the drive shaft 104 and which are mounted within a cylinder block so as to be movable back and forth corresponds to that which is in accordance with the prior art, for example in accordance with DE 197 49 727 A1.

The pivotal mounting of the swash ring 107 defines a pivot axis 101 extending in a transverse direction to the drive shaft 104. The pivot axis 101 is defined, in concrete terms, by two mounting pins mounted coaxially on both sides of the sliding sleeve 108. The mounting pins are mounted in radial bores in the swash ring 107. For the purpose, the sliding sleeve can additionally have, on both sides, mounting sleeves which bridge the annular space between the sliding sleeve 108 and the swash ring 107. This arrangement also corresponds substantially to the prior art in accordance with DE 197 49 727 A1.

Of significance is the axial support of the swash ring at the supporting element 109 arranged to rotate together with the drive shaft 104. That support is provided by means of a supporting arc 110 arranged on the swash ring 107. The supporting arc 110 is so constructed that it overlaps an articulated arrangement effective between the pistons and swash ring, more particularly in such a manner that, irrespective of the inclination of the swash ring 107, the possibility is excluded of a collision between

the swash ring 107 and supporting arc 110, on the one hand, and a bridge-like piston foot encompassing the afore-mentioned articulated arrangement, on the other hand. The supporting element 109 is an integral part of a disc 112 rotating together with the drive shaft 104, more particularly a circle segment of raised construction relative to the disc.

The supporting surface of the arc 110 extends approximately concentrically to the centre-point of the articulated arrangement effective between the pistons and swash plate or swash ring 107, which articulated arrangement includes sliding blocks in the shape of segments of a sphere. The axial support is accordingly effective outside the afore-mentioned articulated arrangement, with the consequence that the articulated arrangement that is effective between the pistons and swash plate or swash ring is not impaired by axial support measures. This is valid especially for the dimensioning of the afore-mentioned articulated arrangement.

It will furthermore be seen that, in the case of the embodiment shown, the pivotal mounting of the swash plate or swash ring 107 serves only for transmitting torque and the supporting element 109 serves only for axially supporting the pistons and/or for gas force support. The transmission of torque is accordingly de-coupled from the axial support of the swash ring 107.

In Figs. 1 - 4, the swash ring is in an inclination position for maximal piston stroke. Figs. 5 - 8 show the swash ring in a position for minimal piston stroke.

The circles additionally indicated in Figs. 4 and 8, in continuation of the supporting surface of the supporting arc 110, show that the supporting surface of the supporting arc 110 describes an arc of a circle. If so required, it is possible to depart therefrom deliberately in order to balance out a predetermined "offset" of the support of the supporting arc 109 from the piston longitudinal axis in the event of a change in the inclination of the swash ring 107.

The supporting arc 110 either can be an integral part of the swash ring 107 or, in accordance with Figs. 3 and 7, can be in the form of a separate part rigidly connected to the swash ring 107. The latter arrangement has the advantage that the swash ring can be accurately ground on both flat faces resulting in correspondingly high parallelism of the two contact surfaces located opposite one another for the above-mentioned sliding blocks of an articulated arrangement effective between the

pistons and swash ring.

If the supporting arc 110 is also intended to serve for torque transmission, it preferably extends into a corresponding trough on that face of the supporting element 109 which faces the supporting arc 110. The trough is then preferably in the form of a radial groove.

It should be pointed out again at this juncture that the described arrangement of a swash plate mechanism is given only by way of example. The concept according to the invention is just as suitable, for example, for a swash plate or swash ring mechanism according to DE 197 49 727 A1.

The swash ring 107 is preferably balanced, more particularly in such a manner that the centre of gravity is located at the so-called tilting point. For the purpose, a balancing weight 114 can also be provided diametrically opposite the supporting arc 110 relative to the drive shaft 104, as is shown - merely by way of example - in Fig. 3.

As already mentioned at the beginning, the mean radius governed by the geometry and/or by the density distribution and/or the mean height of the swash plate or of the pivotal portion thereof is/are so selected that the centrifugal forces occurring on rotation of the swash ring are sufficient to counteract the pivotal movement of the swash ring to provide deliberate regulation and thereby to influence, especially to reduce or to limit, the piston stroke and, consequently, the quantity delivered.

In the case of the arrangement shown, the swash plate is in the form of a swash ring. In addition, it can be advantageous to provide - for an articulated connection to other components of the drive mechanism or for balancing of masses - deformations, bores, projections or the like. At any event, the mass centre of gravity should preferably coincide with the tilting point (tilting articulation) of the swash ring.

The inner and outer diameters of the swash ring 107 are governed by the diameters of the sliding blocks, which are part of an articulated arrangement effective between the pistons and swash ring. The afore-mentioned diameters are so selected that the sliding blocks are substantially in contact with the flat sides of the swash ring, more particularly in such a manner that, even in the case of an extreme inclination of the swash ring, they extend only very slightly beyond the outer or inner diameter of the

swash ring. At any event, under the prevailing circumstances, both the inner and also the outer radius of the swash ring should be maximal, the outer diameter of course also being limited by the inner diameter of the housing, which bounds the drive mechanism space.

The mass of the afore-mentioned supporting arc 110 is negligible compared to the other parts of the swash ring. It needs to be taken into account only in respect of possible imbalances, for example by arranging counter-weights having a compensatory action.

The pistons which are used in the drive mechanism according to the invention have a mass of about 30 g to 90 g, preferably 35 g to 50 g. To that end, they are made from aluminium or an aluminium alloy (with or without plastics coating) or from a plastics composite. The use of steel, cast steel or grey cast iron for the pistons is also feasible. The consequence then is, of course, that the piston masses become larger. As a compromise, a combination of steel and aluminium is worthy of consideration. A combination of metal and plastics is also feasible.

Usually, the inner radius "r_i" of the swash ring 107 is in the range from 12 mm to 22 mm. The outer radius "r_a" of the swash ring 107 is about 34 mm to 42 mm.

The pistons are located on a pitch circle diameter "r_m" in the range between 24 mm and 34 mm.

Preference is given to a geometry in the region of "r_i = 20 mm", "r_m = 29 mm" and "r_a = 38 mm", r_m being calculated from the equation $r_m = (r_a + r_i)/2$.

The height "h" of the swash ring 107 is in the range from 8 mm to 20 mm, preferably in the range between 14 mm and 16 mm.

The material used to make the swash ring 107 should preferably have a density greater than 7 g/cm^3, especially greater than 8 g/cm^3.

The swash ring preferably consists of at least two materials in order to obtain optimum mass inertia. Figs. 3 and 7 show a compound swash ring of such a kind in diagrammatic form, 107i denoting the inner ring and 107a denoting the outer ring. The outer ring 107a preferably consists of a material of relatively high density. In that

respect, Fig. 10 shows an alternative arrangement, which is distinguished by the fact that the outer partial ring 107a of heavy material, that is to say material of relatively high density, for example lead or the like, is located within an outer circumferential groove 113 of the inner partial ring 107i, which is made, for example, from wear-resistant steel. This ensures that the two flat faces of the swash plate, on which faces the sliding blocks of the piston articulation slide, are wear-resistant. Otherwise, steel has a lower density than lead, that is to say the inner partial ring 107i consists of a lighter material than the outer partial ring 107a.

The swash ring 107 preferably has a mass moment of inertia $J_2 = J\eta$ or $J = m/4\ (r_a^2 + r_i^2 + h^2/3)$ which is greater than 100,000 gmm^2. Preferably, the mass moment of inertia is greater than J = 200,000 - 250,000 gmm^2.

Furthermore, the swash ring preferably has a mass moment of inertia of $J_3 = J_\zeta = \frac{m}{2}$ $(r_a^2 + r_i^2)$ which is greater than 200,000 gmm^2, preferably about 400,000 - 500,000 gmm^2.

(Note: Usually (plate or ring) J_δ is always approximately $J_3 = 2 \times J_2$. However, it is primarily J_3 that is important, although J_2 and J_3 are, as described, dependent on one another.)

As mentioned hereinbefore, there are various influencing variables (moments) which affect the regulatory behaviour of the swash plate or swash ring, the objective being to compensate or, where appropriate, over-compensate the moment due to translational masses directly by the moment due to rotating masses.

Hereinbelow there is described the derivation of the so-called moment of deviation, which governs the tilting of the swash plate or swash ring and which, more particularly, in the case shown is solely responsible for the tilting of the swash plate or swash ring provided that the mass centre of gravity of the swash plate or swash ring is located both at the tilting point and also at the geometric centre-point of the swash plate or swash ring. This represents an ideal case of the arrangement that is to be aspired to. For the derivation of the moment of deviation the following very generally applies, with reference to Fig. 9:

$$J_{yz} = -J_1 \cos\alpha_2 \cos\alpha_3 - J_2 \cos\beta_2 \cos\beta_3 - J_3 \cos\gamma_2 \cos\gamma_3$$

$$\left.\begin{aligned} \alpha_1 &= 0 \\ \beta_1 &= 90° \\ \gamma_1 &= 90° \end{aligned}\right\} \text{ Direction angles of the x axis relative to the main inertia axes } \xi\text{-}\eta\text{-}\zeta$$

$$\left.\begin{aligned} \alpha_2 &= 90° \\ \beta_2 &= \psi \\ \gamma_2 &= 90° + \psi \end{aligned}\right\} \text{ Direction angles of the y axis relative to the main inertia axes } \xi\text{-}\eta\text{-}\zeta$$

$$\left.\begin{aligned} \alpha_3 &= 90° \\ \beta_3 &= 90° - \psi \\ \gamma_3 &= \psi \end{aligned}\right\} \text{ Direction angles of the z axis relative to the main inertia axes } \xi\text{-}\eta\text{-}\zeta$$

$$J_2 = J_\eta = \frac{m}{4} \left(r_a^2 + r_i^2 + \frac{h^2}{3}\right)$$

$$J_3 = J_\zeta = \frac{m}{2} \left(r_a^2 + r_i^2\right)$$

(Note: $J_3 \approx 2\, J_2$.

Aim: J_{yz} should have a particular magnitude

$J_{yz} \uparrow \} J_3 \uparrow J_2$ necessarily increases!)

Moment of deviation

$$J_{yz} = -J_2 \cos\psi \sin\psi + J_3 \cos\psi \sin\psi$$

Otherwise the following holds true independently of Fig. 9:

<u>Moment due to mass force of the pistons</u>

$$\beta_i = \theta + 2\pi\,(i-1) \cdot \frac{1}{n}$$

$$z_i = R \cdot \omega^2 \tan\alpha \cos\beta_i$$

$$F_{mi} = m_k \cdot z_i$$

$$M(F_{mi}) = m_k \cdot R \cdot \cos\beta_i \cdot z_i$$

$$M_{k,ges} = m_k \cdot R \sum_{i=1}^{n} z_i \cdot \cos\beta_i$$

<u>Moment due to moment of deviation Msw</u>

$$M_{sw} = J_{yz} \cdot \omega^2$$

$$J_{yz} = \left\{\frac{msw}{2}\left(r_a^2 + r_i^2\right) - \frac{msw}{4}\left(r_a^2 + r_i^2 + \frac{h^2}{3}\right)\right\} \cos\alpha \sin\alpha$$

$$J_{yz} = \frac{msw}{24} \sin 2\alpha \left(3r_a^2 + 3r_i^2 - h^2\right)$$

The variables used above have the following meanings:

θ	rotation angle of the shaft (the considerations above and below being made on the basis of $\theta = 0$ for the sake of simplicity)
η	number of pistons
R	distance from piston axis to shaft axis
ω	speed of rotation of shaft
α	tilting angle of the swash ring/swash plate
mk	mass of a piston including sliding blocks or pair of sliding blocks
mk,ges	mass of all pistons including sliding blocks
msw	mass of swash ring
ra	outer radius of swash ring
ri	inner radius of swash ring
h	height of swash ring
ρ	density of swash ring
V	volume of swash ring
βi	angle position of piston i
zi	acceleration of piston i
Fmi	mass force of piston i (including a pair of sliding blocks)
M(Fmi)	moment due to mass force of piston i
Mk,ges	moment due to mass force of all pistons
Msw	moment due to righting moment of the swash ring/swash plate (moment of deviation)

In the context of the present invention it is an objective that the moment due to the righting moment of the swash plate or swash ring, that is to say the moment of deviation, should be greater than/equal to the moment due to the mass forces of all the pistons, that is to say that the following relation should apply:

$$M_{sw} \geq Mk,ges$$

or

$$\left[\omega^2 \frac{msw}{24} \sin 2\alpha \left(3r_a^2 + 3r_i^2 - h^2\right) \geq \omega^2 R^2 \; m_k \tan\alpha \sum_{i=1}^{n} \cos^2\beta i\right]$$

The above equation shows that the speed of rotation has an influence on both terms equally and therefore changes in the speed of rotation do not affect the moment ratio. This can also be seen from the following example in Tables 1 and 2, wherein it is assumed that:

number of pistons: $n = 7$
distance of piston axis to drive shaft longitudinal axis: $R = 25$ mm
inner radius r_i/outer radius r_a of swash ring: $r_i/r_a = 15/35$
density: $\rho = 7.9$
height of swash ring: $h = 10$ mm
mass/piston: $m_k = 39$

Table 1

Influence of the speed of rotation on the tilting moments			
n	Mk,ges	Msw	alpha
[rev./min.]	[Nm]	[Nm]	[°]
800	0.11	0.11	10
1500	0.37	0.37	10
3000	1.48	1.48	10
5000	4.12	4.12	10
8000	10.56	10.55	10
11000	19.98	19.95	10

Table 2

Influence of the tilting angle on the tilting moments			
alpha	Mk,ges	Msw	n
[°]	[Nm]	[Nm]	[rev./min.]
0	0.00	0.00	3000
3	0.44	0.45	3000
6	0.88	0.90	3000
10	1.48	1.48	3000
14	2.10	2.04	3000
18	2.74	2.55	3000

Table 2 shows that the particular swash plate/ring tilting angle changes the moment ratio very little. Moreover from the above equation it can be seen that $\tan\alpha \neq \sin 2\alpha$ and that consequently the moment ratio is of low dependence on the angle α, especially for small angles α. That results in sensible dimensioning for a medium angle α: $M_{k,ges} = M_{sw}$, or for α_{max}: $M_{k,ges} = M_{sw}$. The swash plate then has a compensating action.

Table 3 below shows results for the following cases:

- The swash plate has a compensating action.
- The swash plate has a down-regulating or over-compensating action.

Table 3

Influence of the swash plate geometry ϱ = 7.9 g/mm³										
ra	ri	h	Jy/mk,ges	Jy/msw	Jz/mk,ges	Jz/msw	msw/mk,ges	Mk,ges	Msw	
35.00	15.00	10.00	337	371	659	725	0.91	1.48	1.48	compensating
37.50	17.50	10.00	436	436	856	856	1.00	1.80	1.93	compensating (~)
40.00	20.00	10.00	555	508	1091	1000	1.09	2.14	2.47	compensating (~)
ra	ri	h	Jy/mk,ges	Jy/msw	Jz/mk,ges	Jz/msw	msw/mk,ges	Mk,ges	Msw	
35.00	15.00	16.00	558	384	1055	725	1.45	1.48	2.29	down-regulating
37.50	17.50	16.00	719	449	1370	856	1.60	1.80	3.00	down-regulating
40.00	20.00	16.00	910	521	1745	1000	1.75	2.14	3.85	down-regulating

wherein "(~)" means the equivalent of "approximately" or "about".

For the moment equilibrium, therefore, substantially only the geometric variables are relevant, although of course the masses of the pistons and of the swash plate also have an influence. In specific terms, the following variables are of relevance to the moment equilibrium:

$$[m_{k,ges} / msw / R^2 / r_a^2 / r_i^2 / h^2]$$

When the swash plate is in the form of a swash ring it is advantageous for the distance between the piston axis and the drive shaft axis to be calculated from the relation:

$$R = (r_a + r_i)/2$$

For optimum dimensioning of the swash plate, in this case of the swash ring 107, reference is preferably made to the quotient "moment of inertia/mass", that is to say "J/m". This quotient expresses the magnitude of the mass moment of inertia for a predetermined swash plate/ring mass. The quotient should be greater than 250 gmm^2/g. Quotients greater than 400 - 500 gmm^2/g are especially advantageous. "J" refers to each centre of gravity axis (that is to say: $J = J_1 = J_2 = J_3$, it being generally true that $J_3 \approx 2J_2$), the centre of gravity of the rotating mass preferably being located in the centre of the tilt-providing articulation thereof.

Relatively large mass inertias should be selected especially when piston masses substantially greater than 40 g/piston are selected.

By means of the arrangement according to the invention there should be achieved above all a mechanical down-regulation of the quantity delivered by an axial piston compressor in the event of an increase in the speed of rotation. The ideal case would of course be constant regulation, constant regulation being a sub-case of the mechanical down-regulation resulting from geometry and moment distribution that is desired in accordance with the invention.

The following example shows advantageous dimensioning for various radii, volumes and masses for a swash ring 107:

Table 4

Inner radius r_i [mm]	Outer radius r_a [mm]	Height H [mm]	Density ς [g/cm3]	Volume V [mm^3]	Mass m [g]	Mass inertia J [g mm^2]	Quotient [g mm^2/g]
15.0	35.0	10.0	7.9	31416	248	92036	371
17.5	37.5	10.0	7.9	34558	273	119155	437
20.0	40.0	10.0	7.9	37699	298	151393	508
15.0	35.0	16.0	7.9	50265	397	152419	384
17.5	37.5	16.0	7.9	55292	437	196327	449
20.0	40.0	16.0	7.9	60319	477	248424	521

The above values are obtained from the following generally valid equations for a swash ring:

(1) $m = \varsigma V$

(2) $V = \frac{\pi}{4} \cdot h\,(D^2 - d^2)$

(3) $D = 2r_a$

(4) $d = 2r_i$

(5) $J = m/4\,(r_a^2 + r_i^2 + h^2/3)$

As described, reference is made very generally, in dimensioning, preferably to the quotient "J/m" and also, preferably, specifically to the ratio "$J_y/m_{K,ges}$", that is to say to the quotient of the mass inertia of the swash plate or swash ring in relation to the y axis according to Fig. 9 and the total piston masses. That quotient can be used alternatively to the above-mentioned measures or in parallel therewith for dimensioning the arrangement and accordingly for obtaining a desired regulatory behaviour.

In the process it can be assumed that a ratio of rotating masses to translational masses $m_{sw}/m_{k,ges}$ of much less than "1" has a very disadvantageous effect on the regulatory behaviour desired herein. The afore-mentioned ratio must therefore be avoided.

In the case of a mass ratio of $m_{sw}/m_{k,ges} = 1$ there is preferably obtained for the ratio $J_y/m_{k,ges}$ a minimum value for compensation of about 250 ... 300 g mm^2/g.

Higher values can be set depending on the desired regulatory behaviour; of particular interest, however, is exact compensation of changes in the swash plate tilting angle in the case of a mass ratio of $m_{k,ges} = m_{sw}$.

Over-compensation can also be of interest, especially in the case of compensation of the change in the quantity delivered as a consequence of changes in the speed of rotation.

In analogous manner, the quotients $J_z/m_{k,ges}$ and J_z/m_{sw} can also be used for dimensioning for the desired regulatory behaviour, because the moment of inertia J_z in relation to the z or drive shaft axis, together with J_y, forms the governing moment of deviation. For the described swash ring geometry the following relation applies:

$$J_z \approx 2 J_y.$$

Because $J_{yz} \approx J_z (\ldots) - J_y (\ldots)$ and because J_{y2} should be large, J_z is actually the more important variable. J_y can be used as the reference variable solely for the reason that the afore-mentioned relation $J_z \approx 2 J_y$ applies.

All features disclosed in the application documents are claimed as being important to the invention insofar as they are novel on their own or in combination compared with the prior art.

Reference numerals

100	swash plate mechanism
101	pivotal mounting (pivotal axis)
104	drive shaft
107	swash ring
107i	inner swash ring
107a	outer swash ring
108	sliding sleeve
109	supporting element (axial support)
110	supporting arc
112	disc
113	circumferential groove
114	balancing weight